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DESCRIPTION

APPARATUS FOR CALCULATING AMOUNT OF RECIRCULATED
EXHAUST GAS FOR INTERNAL COMBUSTION ENGINE

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FIELD OF THE INVENTION

The present invention relates to an apparatus for calculating an amount of a recirculated exhaust gas for an internal combustion engine.

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BACKGROUND ART

Conventionally, there is known an internal combustion engine in which an intake pipe downstream of a throttle valve and an exhaust pipe are connected with each other via an exhaust gas recirculation passage and 15 an exhaust gas recirculation control valve, for controlling an amount of recirculated exhaust gas flowing through the exhaust gas recirculation passage, is disposed in the exhaust gas recirculation passage.

20 In such an engine, in order to make an air-fuel ratio accurately equal to a target air-fuel ratio, it is necessary to obtain an amount of fresh air charged in a cylinder, i.e., a *cylinder-charged air amount*, accurately.

25 Thus, there is publicly known an internal combustion engine in which a model is built while taking account of both an amount of fresh air passing through the throttle valve and then flowing into the intake pipe, i.e., a *throttle valve passing-through air amount*, and an amount 30 of the recirculated exhaust gas passing through the exhaust gas recirculation control valve and flowing into the intake pipe, i.e., an *exhaust gas recirculation control valve passing-through gas amount*, and the *cylinder-charged air amount* is calculated using this model (see Japanese Unexamined Patent Publication (Kokai) 35 No. 2002-147279).

35 However, in this Publication, there is no specific description regarding how to obtain the amount of the

recirculated exhaust gas passing through the exhaust gas recirculation control valve.

DISCLOSURE OF THE INVENTION

Therefore, it is an object of the present invention
5 to provide an apparatus for calculating an amount of a recirculated exhaust gas for an internal combustion engine which can provide the exhaust gas recirculation control valve passing-through gas amount, simply and accurately.

10 According to the present invention, there is provided an apparatus for calculating an amount of a recirculated exhaust gas for an internal combustion engine, in which an intake pipe downstream of an throttle valve and an exhaust pipe are connected with each other via an exhaust gas recirculation passage, and an exhaust gas recirculation control valve for controlling an amount of recirculated exhaust gas flowing through the exhaust gas recirculation passage is disposed in the exhaust gas recirculation passage, the apparatus comprising: means
15 for expressing a difference between a cylinder-charged air amount in a steady engine operation with the recirculated exhaust gas being not supplied, and the cylinder-charged air amount in the engine steady operation with the recirculated exhaust gas being supplied, with a function expression of an intake pipe pressure, and for obtaining and storing the function expression in advance, the cylinder-charged air amount being an amount of fresh air charged in a cylinder, and the intake pipe pressure being a pressure in the intake pipe downstream of the throttle valve; means for
20 obtaining the intake pipe pressure; and means for calculating the difference from the obtained intake pipe pressure using the function expression, and for calculating an exhaust gas recirculation control valve
25 passing-through gas amount, which is an amount of the recirculated exhaust gas passing through the exhaust gas recirculation control valve when the exhaust gas
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recirculation control valve is opened, based on the difference.

5 The present invention may be more fully understood from the description of the preferred embodiments of the invention as set forth below, together with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig. 1 shows a general view of an internal combustion engine; Figs. 2A and 2B show diagrams for explaining *EGR control valve passing-through gas flow rate megr*; Fig. 3 shows a diagram illustrating an exhaust pressure P_e , an exhaust temperature T_e and $P_e/\sqrt{T_e}$; Figs. 4A and 4B show diagrams illustrating a function $\Phi(P_m/P_e)$; Fig. 5 shows a diagram illustrating an example of a relationship between an engine load ratio K_{L0n} and an *intake pipe pressure* P_m ; Figs. 6A and 6C show diagrams illustrating gradient e_1 ; Figs. 6B and 6D show diagrams illustrating gradient e_2 ; Fig. 7 shows a diagram illustrating an *intake pipe pressure* b at a connecting point; Figs. 8A and 8B show diagrams illustrating an engine load ratio r at a connecting point; Fig. 9 is a diagram illustrating an example of a relationship between the engine load ratio K_{L0n} and the *intake pipe pressure* P_m ; Fig. 10 is a diagram illustrating an example of a relationship between an engine load ratio K_{L0ff} and the *intake pipe pressure* P_m ; Figs. 11A and 11B show diagrams illustrating gradients a_1 and a_2 , respectively; Fig. 12 shows a diagram illustrating an engine load ratio c at a connecting point; Fig. 13 shows a diagram for explaining a difference ΔK_L ; Fig. 14 shows a flowchart illustrating a calculation routine of the *EGR control valve passing-through gas flow rate megr*; Fig. 15 shows a diagram illustrating an example of a relationship between the difference ΔK_L and the *intake pipe pressure* P_m ; Figs. 16A through 16C show diagrams illustrating gradients h_1 and h_2 and a difference i at a connecting point,

respectively; Fig. 17 shows a flowchart illustrating a calculation routine of the *EGR control valve passing-through gas flow rate megr* according to another embodiment of the present invention; Fig. 18 shows a 5 diagram illustrating a relationship between an opening degree of the *EGR control valve* and a step number *STP*; Figs. 19A through 19C show diagrams illustrating various correction coefficients, respectively; Fig. 20 shows a partial view of an internal combustion engine 10 illustrating yet another embodiment of the present invention; Figs. 21A through 21 C show partial views of different internal combustion engines to which the present invention can be applied; Fig. 22A and 22B show diagrams illustrating another embodiment of the present 15 invention; Fig. 23 shows a flowchart illustrating a calculation routine of *The EGR control valve passing-through gas flow rate megr* according to another embodiment of the present invention.

BEST MODE FOR CARRYING OUT THE INVENTION

20 Fig. 1 shows a case in which the present invention is applied to a spark ignition internal combustion engine. However, the present invention can also be applied to a compression ignition internal combustion engine.

25 With reference to Fig. 1, numeral 1 depicts an engine body 1 comprising, for example, four cylinders, 2 depicts a cylinder block, 3 depicts a cylinder head, 4 depicts a piston, 5 depicts a combustion chamber, 6 depicts an intake valve, 7 depicts an intake port, 8 depicts an exhaust valve, 9 depicts an exhaust port 9, 10 depicts an ignition plug, and 11 depicts a fuel injector. The intake port 7 is coupled to a surge tank 13 via a respective intake branch 12 and the surge tank 13 is, in turn, coupled to an air cleaner 15 via an intake duct 14. 30 A throttle valve 17 driven by a step motor 16 is disposed in the intake duct 14. Note that the intake duct 35 downstream of the throttle valve 17, the surge tank 13,

the intake branch 12 and the intake port 7 may be collectively referred to as an intake pipe in this specification.

On the other hand, the exhaust port 11 is coupled to 5 a catalytic converter 20 via an exhaust manifold 18 and an exhaust pipe 19, and the catalytic converter 20 is communicated to an atmosphere via a muffler (not shown).

The exhaust manifold 18 and each intake branch 12 are coupled to each other via an exhaust gas 10 recirculation (hereinafter referred to as EGR) supply pipe 21 and an electrically-controlled EGR control valve 22 is disposed in the EGR supply pipe 21. In the 15 internal combustion engine shown in Fig. 1, the EGR supply pipe 21 downstream of the EGR control valve 22 is split into branches connected to the respective intake branch 12. Here, the EGR control valve 22 is provided 20 with a step motor and as a step number STP of this step motor is increased, an opening degree of the EGR control valve 22 is also increased. In other words, the step number STP represents the opening degree of the EGR control valve 22.

An electronic control unit 30 consists of a digital computer and comprises a ROM (read only memory) 32, a RAM (random access memory) 33, a CPU (microprocessor) 34, an 25 input port 35 and an output port 36, which are interconnected by a bidirectional bus 31. A pressure sensor 39 for detecting an intake pipe pressure P_m , which is a pressure in the intake pipe, is attached to the surge tank 13. Further, a throttle opening degree sensor 30 40 for detecting an opening degree of the throttle valve is attached to the throttle valve 17. Still further, a load sensor 42 for detecting a depression of an accelerator pedal 41 is connected to the accelerator pedal 41. The depression of the accelerator pedal 41 35 represents a required load. Still further, an atmospheric temperature sensor 44 for detecting an atmospheric temperature and an atmospheric pressure

sensor 45 for detecting an atmospheric pressure are attached to the intake duct 14 and a water temperature sensor 46 for detecting an engine coolant temperature THW is attached to the cylinder block 2. Output signals of these sensors 39, 40, 42, 44, 45 and 46 are input to the input port 35 via respective AD converters 37. Still further, a crank angle sensor 43 generating an output pulse at every 30° rotation of a crank shaft, for example, is connected to the input port 35. The CPU 34 calculates an engine speed NE based on the output pulse of the crank angle sensor 43. On the other hand, the output port 36 is connected via respective driving circuits 38 to the ignition plug 10, the fuel injector 11, the step motor 16 and the EGR control valve 22, which are controlled based on an output signal from the electronic control unit 30.

In the internal combustion engine shown in Fig. 1, a fuel injection amount QF is calculated based on the following equation, for example:

$$QF = kAF \cdot KL$$

where kAF represents an air-fuel ratio setting coefficient and KL represents an engine load ratio (%).

The air-fuel ratio setting coefficient kAF is a coefficient representing a target air-fuel ratio which becomes small as the target air-fuel ratio becomes large or lean and which becomes large as the target air-fuel ratio becomes small or rich. The air-fuel ratio setting coefficient kAF is stored in the ROM 32, in advance, as a function of an engine operating condition such as the required load and the engine speed.

On the other hand, the engine load ratio KL represents an amount of fresh air charged in each cylinder and is defined by the following equation, for example:

$$KL(\%) = \frac{M_{\text{cair}}}{\frac{DSP}{NCYL} \cdot p_{\text{astd}}} \cdot 100$$
$$= kk \cdot M_{\text{cair}}$$

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Here, M_{cair} represents a cylinder-charged air amount (g) which is an amount of fresh air having been charged in each cylinder when the intake valve 7 is closed after it is opened, DSP represents an engine displacement (liter), NCYL represents the number of cylinders, and p_{astd} (about 1.2 g/liter) represents a density of air in standard conditions (1 atmospheric pressure, 25°C). Further, kk represents these coefficients integrated into one value, and thus the cylinder-charged air amount M_{cair} can be expressed by KL/kk .

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Therefore, what is needed to make an actual air-fuel ratio equal to the target air-fuel ratio accurately, is to obtain the engine load ratio KL accurately.

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When the EGR control valve 22 is opened and thus the EGR gas is supplied, a gas mixture of the fresh air and the EGR gas is sucked into each cylinder. Therefore, if the amounts of the gas mixture and the EGR gas having been charged in each cylinder when the intake valve 7 is closed after it is opened, are referred to as a cylinder-charged gas amount Mc and a cylinder-charged EGR gas amount M_{cegr} , respectively, the cylinder-charged gas amount Mc can be expressed with a sum of the cylinder-charged air amount M_{cair} and the cylinder-charged EGR gas amount M_{cegr} ($Mc = M_{\text{cair}} + M_{\text{cegr}}$).

In this connection, it is known that a cylinder-charged gas amount Mc can be expressed with a linear function expression of the intake pipe pressure P_{in} when the intake valve 7 is closed. That is, theoretically and empirically, the cylinder-charged gas amount Mc is proportional to a pressure in a cylinder when the intake valve 7 is closed and this pressure in the cylinder is substantially equal to a mixed gas pressure upstream of

the intake valve 7, or the intake pipe pressure P_m .

As only fresh air is charged in the cylinder when the EGR gas is not supplied, the cylinder-charged air amount M_{cair} and thus the engine load ratio KL can be expressed with the linear function expression of the intake pipe pressure P_m at this condition. Therefore, the engine load ratio KL can be obtained simply and accurately.

However, when the EGR gas is supplied, the circumstances are completely different in that not only the fresh air but also the EGR gas is charged in the cylinder. Therefore, conventionally, it has been thought absolutely impossible to express the cylinder-charged air amount M_{cair} or the engine load ratio KL with the linear function expression of the intake pipe pressure P_m .

Considering the fact that the cylinder-charged gas amount M_c can be expressed with the linear function expression of the intake pipe pressure P_m and is a sum of the cylinder-charged air amount M_{cair} and the cylinder-charged EGR gas amount M_{cegr} , the cylinder-charged air amount M_{cair} or the engine load ratio KL when the EGR gas is supplied can be expressed with the linear function expression of the intake pipe pressure P_m , if the cylinder-charged EGR gas amount M_{cegr} can be expressed with the linear function expression of the intake pipe pressure P_m .

However, conventionally, it has been thought impossible to express the cylinder-charged EGR gas amount M_{cegr} with the linear function of the intake pipe pressure P_m , too. This will be described with reference to Figs. 2A and 2B.

First, as shown in Fig. 2A, assuming that an EGR gas pressure upstream of the EGR control valve 22 is equal to an exhaust pressure P_e (kPa) in the exhaust manifold 18, a EGR gas temperature upstream of the EGR control valve is equal to an exhaust gas temperature T_e (K) in the exhaust manifold 18, and a pressure of the EGR gas

5 passing through the EGR control valve 22 is the intake pipe pressure P_m (kPa), an EGR control valve passing-through gas flow rate meg_r (g/sec), which is a flow rate of the EGR gas passing through the EGR control valve 22, can be expressed with the following equation (1):

$$meg_r = \mu \cdot A_e \cdot \frac{P_e}{\sqrt{R_e \cdot T_e}} \cdot \Phi\left(\frac{P_m}{P_e}\right) \quad \dots (1)$$

10 Here, μ represents a flow rate coefficient at the EGR control valve 22, A_e represents a cross sectional area of an opening of the EGR control valve 22 (m^2), R_e represents a constant regarding the gas constant R , and $\Phi(P_m/P_e)$ represents a function of P_m/P_e . Here, the flow rate coefficient μ and the opening cross sectional area A_e depend on an opening degree θ_e of the EGR control valve 22, and the constant R_e is obtained by dividing the gas constant R by a mass M_e of the exhaust gas or the EGR gas per 1 mol ($R_e = R/M_e$).

15 Further, the function $\Phi(P_m/P_e)$ is expressed with the following equation using a specific heat ratio κ (constant) so that the flow rate of the EGR gas does not exceed a sonic velocity:

$$20 \Phi\left(\frac{P_m}{P_e}\right) = \begin{cases} \sqrt{\frac{\kappa}{2(\kappa+1)}} & \dots \frac{P_m}{P_e} > \frac{1}{\kappa+1} \\ \sqrt{\left(\frac{\kappa-1}{2\kappa}\right) \cdot \left(1 - \frac{P_m}{P_e}\right) + \frac{P_m}{P_e}} \cdot \left(1 - \frac{P_m}{P_e}\right) & \dots \frac{P_m}{P_e} \leq \frac{1}{\kappa+1} \end{cases}$$

25 Briefly described, the equation (1) mentioned above is derived using conservation laws of mass, energy and momentum regarding the EGR gas at the upstream and downstream of the EGR control valve 22, as well as the characteristic equations of the EGR gas at the upstream and downstream of the EGR control valve 22.

30 35 Here, assuming that the exhaust pressure P_e is equal

to the atmospheric pressure P_a in order to simplify calculation, the *EGR control valve passing-through gas flow rate megr* expressed with the equation (1) appears as shown in Fig. 2B. More specifically, when the *intake pipe pressure P_m* is low, the *EGR control valve passing-through gas flow rate megr* is maintained substantially constant and, as the *intake pipe pressure P_m* is increased, the *EGR control valve passing-through gas flow rate megr* is reduced toward the atmospheric pressure while showing nonlinearity to the *intake pipe pressure P_m* , as shown by NR in Fig. 2B. Here, this nonlinear portion NR is based on the term $P_e/\sqrt{T_e}$ and the function $\Phi(P_m/P_e)$ in the equation (1).

Therefore, it has been thought impossible to express the *EGR control valve passing-through gas flow rate megr* and, in particular, its nonlinear portion NR with a linear function expression of the *intake pipe pressure P_m* . Indeed, if a considerably large number of linear function expressions of the *intake pipe pressure P_m* are used, it may be thought possible to express the *EGR control valve passing-through gas flow rate megr* with the linear function expressions. However, in this case, it cannot be said that the engine load ratio KL is obtained simply.

However, the inventors of the present invention have found that the *EGR control valve passing-through gas flow rate megr* can be expressed with two linear function expressions of the *intake pipe pressure P_m* and, therefore, the *cylinder-charged air amount M_{air}* or the engine load ratio KL can also be expressed with two linear function expressions of the *intake pipe pressure P_m* .

Specifically, first, as shown in Fig. 3, as the *intake pipe pressure P_m* increases, the *exhaust gas temperature T_e* increases more significantly than the *exhaust pressure P_e* increases and, as a result, $P_e/\sqrt{T_e}$

can be expressed by the linear function expression of the intake pipe pressure P_m .

Further, the function $\Phi(P_m/P_e)$ can also be expressed with the linear function expression of the intake pipe pressure P_m . This will be explained with reference to Figs. 4A and 4B. Considering the fact that the exhaust pressure P_e is not maintained at the constant atmospheric pressure P_a but it varies in accordance with the intake pipe pressure P_m , the function $\Phi(P_m/P_e)$ when the intake pipe pressure P_m is equal to P_{m1} lies not on a curve C_A converging to the atmospheric pressure P_a , but on a curve C_1 converging to the exhaust pressure P_{e1} , as shown by plots (O) as shown in Fig. 4A. Similarly, $\Phi(P_m/P_e)$ when $P_m = P_{m2} (> P_{m1})$ lies on a curve C_2 converging to the exhaust pressure $P_{e2} (> P_{e1})$ and, $\Phi(P_m/P_e)$ when $P_m = P_{m3} (> P_{m2})$ lies on a curve C_3 converging to the exhaust pressure $P_{e3} (> P_{e2})$.

The plots obtained in this way can be connected by a straight line L_2 as shown in Fig. 4B. Therefore, the function $\Phi(P_m/P_e)$ can be expressed with one linear function expression of the intake pipe pressure P_m corresponding to a straight line L_1 when the intake pipe pressure P_m is low, and with the other linear function expression of the intake pipe pressure P_m corresponding to the straight line L_2 when the intake pipe pressure P_m is high and, therefore, it can be expressed with two linear function expressions of the intake pipe pressure P_m . Namely, the EGR control valve passing-through gas flow rate m_{egr} can be expressed with the two linear function expressions of the intake pipe pressure P_m .

Here, in the engine steady operation, the EGR control valve passing-through gas flow rate m_{egr} , which is the EGR gas amount flowing into the intake pipe per unit time, is equal to a cylinder-sucked EGR gas amount m_{cegr} (g/sec), which is the EGR gas amount exiting from

the intake pipe and flowing into the cylinder per unit time. Further, the cylinder-charged EGR gas amount M_{cegr} is obtained by multiplying the cylinder-sucked EGR gas amount m_{cegr} by a time period ΔT (sec) required for one intake stroke of each cylinder ($M_{cegr} = m_{cegr} \cdot \Delta T$).

As a result, the cylinder-charged EGR gas amount M_{cegr} in the engine steady operation can be expressed with the linear function expression of the intake pipe pressure P_m .

Therefore, the cylinder-charged air amount M_{cair} or the engine load ratio KL in the engine steady operation with the EGR gas being supplied can be expressed with the two linear function expressions of the intake pipe pressure P_m .

If the engine load ratio KL when the EGR gas is supplied is referred to as KL_{on} , Fig. 5 shows an example of the two linear function expressions of the intake pipe pressure P_m expressing the engine load ratio KL_{on} in the engine steady operation, with the constant engine speed NE and the constant opening degree of the EGR control valve STP . As shown in Fig. 5, the engine load ratio KL_{on} is expressed with the two linear function expressions having different gradients from each other and continuous with each other at a connecting point CP . More specifically, the engine load ratio KL_{on} is expressed with one linear function expression having a gradient e_1 when the intake pipe pressure P_m is low, and is expressed with the other linear function expression having a gradient e_2 when the intake pipe pressure P_m is high.

Here, designating the gradients of the two linear function expressions as e_1 and e_2 , respectively, and the intake pipe pressure and the engine load ratio at the connecting point CP as b and r , respectively, the two linear function expressions can be expressed with the following equations:

$$KLon = e1 \cdot (Pm - b) + r \quad \dots \quad Pm \leq b$$

$$KLon = e2 \cdot (Pm - b) + r \quad \dots \quad Pm > b$$

These equations can be integrated into the following equation (2):

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$$KLon = e \cdot (Pm - b) + r \quad (2)$$

$$e = e1 \quad \dots \quad Pm \leq b$$

$$e = e2 \quad \dots \quad Pm > b$$

10 In the embodiment of the present invention, the two linear function expressions of the *intake pipe pressure* Pm expressing the engine load ratio $KLon$ in the engine steady operation are stored in the ROM 32 in advance in the form of the equation (2). It allows the two linear function expressions to be expressed with three 15 parameters e , b and r . Thus, the number of parameters required to express the two linear function expressions is reduced.

The parameters e , b and r of the equation (2) is calculated in accordance with the following equations:

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$$e1 = e1^* \cdot ktha$$

$$e2 = e2^* \cdot ktha$$

$$b = b^* \cdot ktha \cdot kpa$$

$$r = r^* \cdot ktha \cdot kpa$$

25 where $e1^*$, $e2^*$, b^* and r^* are the gradients and the *intake pipe pressure* and the engine load ratio at the connecting point, respectively, when an engine environmental condition is a predetermined reference 30 environmental condition. While any condition may be used as the reference environmental condition, the standard condition (1 atmospheric pressure, 25°C) is used as the reference environmental condition in the embodiment of the present invention.

35 On the other hand, $ktha$ and kpa represent an atmospheric temperature correction coefficient and an atmospheric pressure correction coefficient, respectively. The atmospheric temperature correction coefficient $ktha$ is for correcting each of the parameters

17 $e1^*$, $e2^*$, b^* and r^* in the reference environmental condition, based on the actual atmospheric temperature detected by the atmospheric temperature sensor 44, and is made equal to 1.0 when the correction is not necessary.
5 Further, the atmospheric pressure correction coefficient kpa is for correcting each of the parameters b^* and r^* in the reference environmental condition, based on the actual atmospheric pressure detected by the atmospheric pressure sensor 45, and is made 1.0 when the correction
10 is not necessary.

11 Therefore, considering the fact that the atmospheric temperature correction coefficient $ktha$ or the atmospheric pressure correction coefficient kpa are representative values representing the actual engine environmental condition, it can be said that the parameters $e1^*$, $e2^*$, b^* and r^* in the reference environmental condition are corrected based on the representative values representing the actual engine environmental condition. Alternatively, it can be
15 considered that the engine load ratio $KLon$ in the reference environmental condition is corrected based on the representative values representing the actual engine environmental condition.
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21 On the other hand, in the embodiment of the present invention, the parameters e^* ($e1^*$ and $e2^*$), b^* and r^* are set in accordance with the opening degree of the EGR control valve STP or the engine speed NE, taking the fact that the opening cross sectional area Ae of the EGR control valve 22 depends on the opening degree of the EGR control valve STP and the engine charging efficiency depends on the engine speed NE, into account.
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26 More specifically, as shown in Fig. 6A, the gradient $e1^*$ becomes larger as the engine speed NE becomes higher when the engine speed NE is low, becomes smaller as the engine speed NE becomes higher when the engine speed NE is high, and becomes larger as the opening degree of the EGR control valve STP becomes larger. The gradient $e2^*$
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becomes larger as the engine speed NE becomes higher when the engine speed NE is low, becomes smaller as the engine speed NE becomes higher when the engine speed NE is high, and becomes larger as the opening degree of the EGR control valve STP becomes larger, as shown in Fig. 6B. 5 These gradients $e1^*$ and $e2^*$ are obtained by experiment and are stored in the ROM 32, in advance, as functions of the engine speed NE and the opening degree of the EGR control valve STP in the form of maps shown in Figs. 6C 10 and 6D, respectively.

On the other hand, as shown in Fig. 7, the *intake pipe pressure* b^* at the connecting point CP becomes smaller as the engine speed NE becomes higher. The *intake pipe pressure* b^* at the connecting point CP is 15 also obtained by experiment, in advance, and stored in the ROM 32 as a function of the engine speed NE in the form of a map as shown in Fig. 7.

Further, as shown in Fig. 8A, the gradient r^* at the connecting point CP becomes larger as the engine speed NE becomes higher when the engine speed NE is low, becomes smaller as the engine speed NE becomes higher when the engine speed NE is high, and becomes smaller as the opening degree of the EGR control valve STP becomes larger. The engine load ratio r^* at the connecting point CP is also obtained by experiments in advance and stored 20 in the ROM 32 as a function of the engine speed NE and the opening degree of the EGR control valve STP in the form of a map as shown in Fig. 8B. 25

Therefore, generally speaking, two linear function 30 expressions of the *intake pipe pressure* P_m expressing the *cylinder-charged air amount* M_{air} or the engine load ratio K_{Lm} in the engine steady operation are obtained and stored in advance, for different opening degrees of the EGR control valve. Further, two linear function 35 expressions of the *intake pipe pressure* P_m expressing the *cylinder-charged air amount* M_{air} or the engine load ratio K_{Lm} in the engine steady operation are obtained

and stored in advance for different engine speeds.

Fig. 9 shows an example of the two linear function expressions of the *intake pipe pressure* P_m expressing the engine load ratio K_{L0n} in the engine steady operation with a constant engine speed NE and various opening degrees of the EGR control valve. Note that a broken line in Fig. 9 represents the engine load ratio K_{L0ff} when the EGR gas is not supplied or the opening degree of the EGR control valve STP is made zero.

On the other hand, as described above, the engine load ratio K_{L0ff} when the EGR gas is not supplied can be expressed with a linear function expression of the *intake pipe pressure* P_m . Fig. 10 shows an example of two linear function expressions of the *intake pipe pressure* P_m expressing the engine load ratio K_{L0ff} in the engine steady operation with a constant engine speed NE . In the embodiment according to the present invention, as shown in Fig. 10, the engine load ratio K_{L0ff} is expressed with the two linear function expressions of the *intake pipe pressure* P_m having different gradients from each other and continuous with each other at a connecting point CP . More specifically, the engine load ratio K_{L0ff} is expressed with one linear function expression having the gradient a_1 when the *intake pipe pressure* P_m is low, and expressed with the other linear function expression having the gradient a_2 when the *intake pipe pressure* P_m is high.

Here, if the gradients of the two linear function expressions are referred to as a_1 and a_2 , respectively, and the *intake pipe pressure* and the engine load ratio at the connecting point CP are referred to as b and c , respectively, these two linear function expressions can be expressed with the following equations:

$$K_{L0ff} = a_1 \cdot (P_m - b) + c \quad \dots P_m \leq b$$

$$K_{L0ff} = a_2 \cdot (P_m - b) + c \quad \dots P_m > b$$

These equations can be integrated into the following

equation (3):

$$KLoft = a \cdot (Pm - b) + c \quad (3)$$

$$a = a1 \quad \dots \quad Pm \leq b$$

$$a = a2 \quad \dots \quad Pm > b$$

5 In the embodiment of the present invention, the two linear function expressions of the *intake pipe pressure* Pm expressing the engine load ratio $KLoft$ in the steady engine operation are stored in the ROM 32 in advance in the form of the equation (3). In this case, the *intake pipe pressure* b at the connecting point CP is identical to the one at the connecting point CP for the engine load ratio $KLon$ described above. Therefore, the number of parameters can be further reduced. Of course, the *intake pipe pressures* at these connecting points CP may be
10 different from each other.
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The parameters a and c of the equation (3) are calculated based on the following equations:

$$a1 = a1^* \cdot ktha$$

$$a2 = a2^* \cdot ktha$$

$$20 \quad c = c^* \cdot ktha \cdot kpa$$

where, $a1^*$ and $a2^*$, and c^* are the gradients and the engine load ratio at the connecting point, respectively, when an engine environmental condition is the predetermined reference environmental condition as
25 described above, or in the standard condition.

Therefore, considering the fact that the atmospheric temperature correction coefficient $ktha$ or the atmospheric pressure correction coefficient kpa are representative values representing the actual engine
30 environmental condition, it can be said that the parameters $a1^*$, $a2^*$ and c^* in the reference environmental condition are corrected based on the representative values representing the actual engine environmental condition. Alternatively, it can be considered that the engine load ratio $KLoft$ in the reference environmental
35 condition is corrected based on the representative values

representing the actual engine environmental condition.

On the other hand, in the embodiment of the present invention, the parameters a^* ($a1^*$ and $a2^*$) and c^* are set in accordance with the engine speed NE , taking the fact 5 that the engine charging efficiency depends on the engine speed NE into account.

More specifically, as shown in Fig. 11A, the gradient $a1^*$ becomes larger as the engine speed NE becomes higher when the engine speed NE is low, and becomes smaller as the engine speed NE becomes higher when the engine speed NE is high. The gradient $a2^*$ becomes larger as the engine speed NE becomes higher when the engine speed NE is low, and becomes smaller as the engine speed NE becomes higher when the engine speed NE is high, as shown in Fig. 11B. These gradients $a1^*$ and $a2^*$ are obtained by experiments and stored in the ROM 32 in advance as a function of the engine speed NE in the form of maps shown in Figs. 11A and 11B, respectively.

Further, as shown in Fig. 12, the engine load ratio c^* at the connecting point CP becomes larger as the engine speed NE becomes higher when the engine speed NE is low, and becomes smaller as the engine speed NE becomes higher when the engine speed NE is high. The engine load ratio c^* at the connecting point CP is also obtained by experiments in advance and stored in the ROM 32 as a function of the engine speed NE in the form of a map as shown in Fig. 12.

Therefore, generally speaking, it can be said that two linear function expressions of the intake pipe pressure Pm expressing the cylinder-charged air amount M_{air} or the engine load ratio K_{loff} in the engine steady operation for different engine speeds NE are determined and stored in advance.

As a result, when the intake pipe pressure Pm is detected by the pressure sensor 39, for example, the engine load ratio K_{lon} or K_{loff} can be obtained accurately and simply using the equation (2) or (3)

described above, from the detected intake pipe pressure P_m , and thus the air-fuel ratio can be made equal to the target air-fuel ratio accurately and simply.

5 The fact that the engine load ratios K_{L0n} and K_{L0ff} can be expressed with the linear function expression of the intake pipe pressure P_m means that there is no need to create respective maps representing the relationships between the engine load ratios K_{L0n} , K_{L0ff} and the intake pipe pressure P_m . Further, it also means that there is no need to solve complicated equations such as differential equations and, therefore, reduces a computation load on the CPU 34.

10 In this connection, as described above, the engine load ratio K_L represents the cylinder-charged air amount M_{cair} ($M_{cair} = K_L/k_k$). Here, considering the fact that only fresh air is charged in the cylinder when the EGR gas is not supplied, it can be considered that the engine load ratio K_{L0ff} when the EGR gas is not supplied represents a total amount of the gas charged in the cylinder at this time, i.e., the cylinder-charged gas amount M_c .

15 Here, considering that the cylinder-charged gas amount M_c does not change whether the EGR gas is supplied or not, it can be said that the engine load ratio K_{L0ff} when the EGR gas is not supplied represents not only the cylinder-charged gas amount M_c when the EGR gas is not supplied but also the cylinder-charged gas amount M_c when the EGR gas is supplied.

20 On the other hand, as described above, the cylinder-charged air amount M_{cair} in the engine steady operation with the EGR gas being supplied is expressed with the engine load ratio K_{L0n} .

25 Accordingly, it can be said that a result of subtraction ΔK_L ($= K_{L0ff} - K_{L0n}$) of the engine load ratio K_{L0n} when the EGR gas is supplied from the engine load ratio K_{L0ff} when the EGR gas is not supplied represents the cylinder-charged EGR gas amount M_{cegr} in

the engine steady operation.

More specifically, as shown in Fig. 13, for example, assuming that $K_{Loff} = K_{Loff1}$ and $K_{Lon} = K_{Lon1}$ when $P_m = P_{m1}$, the cylinder-charged EGR gas amount M_{cegr} in the engine steady operation is expressed with ΔKL ($= K_{Loff1} - K_{Lon1}$).

Therefore, the cylinder-charged EGR gas amount M_{egr} in the engine steady operation can be calculated based on the following equation (4):

10
$$M_{cegr} = k_{egr1} \cdot \Delta KL \quad (4)$$

where k_{egr1} represents a conversion factor from the engine load ratio KL to the cylinder-charged EGR gas amount M_{cegr} .

15 Therefore, if the intake pipe pressure P_m is detected by the pressure sensor 39, for example, the cylinder-charged EGR gas amount M_{cegr} in the engine steady operation can be obtained accurately and simply using the equation (4) described above, from the detected intake pipe pressure P_m .

20 In this connection, in the engine steady operation, the EGR control valve passing-through gas flow rate m_{egr} and the cylinder-sucked EGR gas amount m_{cegr} are equal to each other and the cylinder-charged EGR gas amount M_{cegr} can be expressed with the product of the cylinder-sucked EGR gas amount m_{cegr} and ΔT ($M_{cegr} = m_{cegr} \cdot \Delta T$), as described above.

25 Therefore, it can be said that the difference ΔT mentioned above also represents the EGR control valve passing-through gas flow rate m_{egr} in the engine steady operation.

30 In the embodiment according to the present invention, the EGR control valve passing-through gas flow rate m_{egr} in the engine steady operation is calculated based on the following equation (5):

35
$$m_{egr} = k_{egr2} \cdot \Delta KL \quad (5)$$

where $kegr2$ represents a conversion factor from the engine load ratio KL to the EGR control valve passing-through gas flow rate $megr$.

As described above, the EGR control valve passing-through gas flow rate $megr$ in the engine steady operation is calculated using the above-described equation (5). However, the EGR control valve passing-through gas flow rate $megr$ in an engine transient operation can also be calculated using this equation (5).

More specifically, considering the fact that the EGR control valve passing-through gas flow rate $megr$ greatly depends on the pressure difference between the upstream and downstream of the EGR control valve 22, i.e., the difference between the exhaust pressure Pe and the intake pipe pressure Pm , and that the exhaust pressure Pe and the exhaust temperature Te upstream of the EGR control valve 22 in the engine transient operation is less different from Pe and Te in the engine steady operation, it can be said that the EGR control valve passing-through gas flow rate $megr$ can be determined if the intake pipe pressure Pm is determined.

Therefore, when the intake pipe pressure Pm is detected by the pressure sensor 39, for example, the EGR control valve passing-through gas flow rate $megr$ both in the engine steady and transient operations can be determined accurately and simply using the above-described equation (5) from the detected intake pipe pressure Pm . In this case, the cylinder-charged EGR gas amount $Mcegr$ in the engine steady operation may be calculated either from the EGR control valve passing-through gas flow rate $megr$ in the engine steady operation or from the difference ΔKL using the above-described equation (4).

Fig. 14 shows a calculation routine for the EGR control valve passing-through gas flow rate $megr$ in the above-described embodiment according to the present invention. This routine is executed by interruption

every predetermined time.

Referring to Fig. 14, first, in step 100, the intake pipe pressure P_m , the engine speed NE and the opening degree of the EGR control valve STP are read in. In the 5 following step 101, the atmospheric temperature correction coefficient k_{tha} and the atmospheric pressure correction coefficient k_{pa} are calculated. In the following step 102, the intake pipe pressure b^* and engine load ratio c^* and r^* at the connecting point CP 10 under the reference environmental condition are calculated from the maps of Figs. 7, 8B, and 12. In the following step 103, the parameters b , c and r are calculated by correcting b^* , c^* and r^* using k_{tha} and k_{pa} . In the following step 104, it is judged whether the 15 detected intake pipe pressure P_m is not higher than the intake pipe pressure b at the connecting point. If $P_m \leq b$, the process proceeds to step 105, where a_{1*} and e_{1*} are calculated from the maps of Figs. 6C and 11A. In the following step 106, the gradients a^* and e^* are set to 20 a_{1*} and e_{1*} , respectively. Then, the process proceeds to step 109. In contrast, if $P_m > b$, the process proceeds to step 107, where a_{2*} and e_{2*} are calculated from the maps of Figs. 6D and 11B. In the following step 108, the 25 gradients a^* and e^* are set to a_{2*} and e_{2*} , respectively. Then, the process proceeds to step 109.

In the step 109, the parameters a and e are calculated by correcting a^* and e^* using k_{tha} and k_{pa} . In the following step 110, the engine load ratio K_{loff} is calculated based on the equation (3) ($K_{loff} = a \cdot (P_m - b) + c$). In the following step 111, the engine load ratio K_{lon} is calculated based on the equation (2) ($K_{lon} = e \cdot (P_m - b) + r$). In the following step 112, the 30 difference ΔKL is calculated ($\Delta KL = K_{loff} - K_{lon}$). In the following step 113, the EGR control valve passing-through gas flow rate meg_r is calculated based on the 35 equation (5) ($meg_r = k_{egr2} \cdot \Delta KL$).

5 In the embodiment described above, the engine load ratios K_{Loff} and K_{Lon} are expressed with the respective two linear function expressions. However, the engine load ratios K_{Loff} and K_{Lon} may also be expressed with
5 respective m -th function expressions of n ($n, m = 1, 2$
 ...)

10 Thus, it can be said that, in the embodiment described above, the cylinder-charged air amount or the engine load ratio K_{Loff} in the engine steady operation with the EGR gas being not supplied is expressed with a first function expression which is a function expression of the intake pipe pressure P_m and the first function expression is obtained and stored in advance, the cylinder-charged air amount or the engine load ratio K_{Lon} in the engine steady operation with the EGR gas being supplied is expressed with a second function expression which is a function expression of the intake pipe pressure P_m and the second function expression is obtained and stored in advance, the cylinder-charged air amounts or the engine load ratios K_{Loff} and K_{Lon} are
15 calculated from the intake pipe pressure P_m obtained in advance using the first and second function expressions, respectively, the difference ΔK_L between these calculated cylinder-charged air amounts or the engine load ratios K_{Loff} and K_{Lon} is calculated and, then, the EGR control valve passing-through gas flow rate $meqr$ is
20 calculated based on the difference ΔK_L .
25

30 In addition, generally speaking, it can be said that the difference ΔK_L between the cylinder-charged air amount or the engine load ratio K_{Loff} in the engine steady operation with the EGR gas being not supplied and the cylinder-charged air amount or the engine load ratio K_{Lon} in the engine steady operation with the EGR gas being supplied, is expressed with function expressions of the intake pipe pressure P_m , and the function expressions are obtained and stored in advance, the intake pipe
35

5 pressure P_m is obtained, the difference ΔK_L is calculated from the obtained intake pipe pressure P_m using the function expressions and, then, the EGR control valve passing-through gas flow rate m_{egr} in the engine steady and transient operations and the cylinder-charged EGR gas amount M_{cegr} in the engine steady operation are calculated based on the difference ΔK_L .

10 Next, another embodiment according to the present invention will be explained.

15 The difference ΔK_L described above can be explained using the equations (3) and (2) expressing K_{loff} and K_{lon} , as the following equation:

$$\Delta K_L = K_{loff} - K_{lon} \\ = (a - e) \cdot (P_m - b) + (c - r) \quad (6)$$

20 Here, if substitution $(a - e) = h$ and $(c - r) = i$ are made, the equation (6) will be rewritten as follows:

$$\Delta K_L = h \cdot (P_m - b) + i \quad (7) \\ h = h_1 \quad \dots \quad P_m \leq b \\ h = h_2 \quad \dots \quad P_m > b$$

25 Therefore, as shown in Fig. 15, the difference ΔK_L is expressed with two linear function expressions of the intake pipe pressure P_m having different gradients from each other and continuous with each other at a connecting point CP. More specifically, the difference ΔK_L is expressed with one linear function expression with the gradient h_1 when the intake pipe pressure P_m is low, and expressed with the other linear function expression with the gradient h_2 when the intake pipe pressure P_m is high.

30 In the embodiment of the present invention, the two linear function expressions of the intake pipe pressure P_m expressing the difference ΔK_L are stored in the ROM 32 in the form of the equation (7). This reduces the number of parameters.

The parameters h , b and i in this equation (7) are

calculated based on the following equations:

$$h1 = h1^* \cdot ktha$$

$$h2 = h2^* \cdot ktha$$

$$i = i^* \cdot ktha \cdot kpa$$

5 where $h1^*$ and $h2^*$, and i^* are the gradients and the difference at the connecting point CP, respectively, when the engine environmental condition is the predetermined reference condition. These values $h1^*$, $h2^*$ and i^* are obtained by experiments and stored in the ROM 32, in advance, as a function of the engine speed NE and the opening degree of the EGR control valve STP in the form of maps shown in Figs. 16A, 16B and 16C, respectively. Here, the parameter b is similar to the one in the embodiment described above and, thus, an explanation 10 therefor is omitted.

15

Therefore, generally speaking, it can be said that two linear function expressions of the intake pipe pressure P_m expressing the difference ΔKL for different opening degree of the EGR control valve STP are obtained 20 and stored in advance. Further, it can also be said that two linear function expressions of the intake pipe pressure P_m expressing the difference ΔKL for different engine speeds are obtained and stored in advance.

Fig. 17 shows a calculation routine for the EGR 25 control valve passing-through gas flow rate $meqr$ in the above-described alternative embodiment. This routine is executed by interruption every predetermined time.

Referring to Fig. 17, first, in step 120, the intake pipe pressure P_m , the engine speed NE and the opening 30 degree of the EGR control valve STP are read in. In the following step 121, the atmospheric temperature correction coefficient $ktha$ and the atmospheric pressure correction coefficient kpa are calculated. In the following step 122, the intake pipe pressure b^* and the difference i^* at the connecting point CP under the 35 reference environmental condition are calculated from the

maps of Figs. 7 and 16C. In the following step 123, the parameters b and i are calculated by correcting b^* and i^* using k_{tha} and k_{pa} . In the following step 124, it is determined whether the detected intake pipe pressure P_m is not higher than the intake pipe pressure b at the connecting point. Then, if $P_m \leq b$, the process proceeds to step 125, where h_1^* is calculated from the map of Fig. 16A. In the following step 126, the gradient h^* is set to h_1^* . Then, the process proceeds to step 129. In contrast, if $P_m > b$, the process proceeds to step 127, where h_2^* is calculated from the map of Fig. 16B. In the following step 128, the gradient h^* is set to h_2^* . Then, the process proceeds to step 129.

In the step 129, the parameter h is calculated by correcting h^* using k_{tha} and k_{pa} . In the following step 130, the difference Δ_{KL} is calculated based on the equation (7) ($\Delta_{KL} = h \cdot (P_m - b) + i$). In the following step 131, the EGR control valve passing-through gas flow rate meg_r is calculated based on the equation (5) ($meg_r = k_{egr2} \cdot \Delta_{KL}$).

Here, the opening degree of the EGR control valve STP will be explained briefly. As described above, the opening degree of the EGR control valve is represented by the step number STP of the step motor of the EGR control valve 22 and, thus, the EGR control valve 22 is closed as the step number STP becomes zero and the opening degree of the EGR control valve becomes larger as the step number STP becomes larger.

However, in fact, as shown in Fig. 18, even when the step number STP is increased from zero, the EGR control valve 22 is not opened at once, but it is opened only after the step number STP exceeds STP1. Therefore, the opening degree of the EGR control valve must be expressed with the result of subtraction (STP - STP1) of STP1 from the step number STP.

Further, as there is typically a manufacturing error

in the EGR control valve 22, the actual opening degree of the EGR control valve expressed by the step number STP may deviate from a proper opening degree. Therefore, in the internal combustion engine shown in Fig. 1, a 5 correction coefficient kg for making the actual opening degree of the EGR control valve equal to the proper opening degree is obtained, and is added to the step number STP.

Therefore, the opening degree of the EGR control 10 valve STP will be expressed with the following equation:

$$STP = STP_0 + kg$$

where STP_0 is a step number at which an EGR control valve 22 having a central value of dimensional tolerance begins opening. In the embodiment according to the present 15 invention, the opening degree of the EGR control valve STP thus calculated is used as an argument for the maps.

In this connection, the *EGR control valve passing-through gas flow rate megr* or the *cylinder-charged EGR gas amount Mcegr* in the engine steady operation 20 calculated as described above may be further corrected in consideration of the exhaust temperature T_e .

An explanation for a case in which the *EGR control valve passing-through gas flow rate megr* is corrected will now be given. The *EGR control valve passing-through gas flow rate megr* in this case is expressed with the 25 following equation, for example:

$$meg_r = meg_r \cdot k_wu \cdot k_rtd \cdot k_{inc}$$

where k_wu , k_rtd and k_{inc} represent a correction coefficient at the time of warming-up, a correction coefficient at the time of retardation, and a correction coefficient at the time of increase of fuel supply amount, respectively.

The correction coefficient at the time of warming-up is intended to correct the *EGR control valve passing-through gas flow rate megr* when the warming-up is in 35 process. The exhaust temperature T_e when the warming-up is in process is lower than that when it is completed

and, thus, the EGR control valve passing-through gas flow rate megr (g/sec) increases accordingly. The EGR control valve passing-through gas flow rate megr calculated using the above-described equation (2), (3) or (7) is the value 5 when the warming-up operation is completed and, therefore, it must be corrected.

As shown in Fig. 19A, the correction coefficient at the time of warming-up k_{wu} becomes smaller as an engine coolant temperature THW representing the extent of warming-up becomes higher, and is maintained at 1.0 after the engine coolant temperature THW becomes equal to or be 10 higher than a temperature TWU representing the completion of warming-up. This correction coefficient at the time of warming-up k_{wu} is stored in the ROM 32 in advance in 15 the form of a map shown in Fig. 19A.

On the other hand, the correction coefficient at the time of retardation k_{rttd} is intended to correct the EGR control valve passing-through gas flow rate megr when a retardation correction of the ignition timing is in 20 process. The exhaust temperature Te when the retardation correction is in process is higher than that when it is not in process and, thus, the EGR control valve passing-through gas flow rate megr is reduced accordingly.

As shown in Fig. 19B, the correction coefficient at 25 the time of retardation k_{rttd} is set to 1.0 when the retardation amount RTD is zero, and becomes smaller as the retardation amount RTD becomes larger. This correction coefficient at the time of retardation k_{rttd} is stored in the ROM 32 in advance in the form of a map 30 shown in Fig. 19B.

Further, the correction coefficient at the time of increase of fuel supply amount k_{inc} is intended to correct the EGR control valve passing-through gas flow rate megr when an increasing correction of fuel supply 35 amount is in process. The exhaust temperature Te when the increasing correction of fuel supply amount is in process is lower than that when it is not in process and,

thus, the *EGR control valve passing-through gas flow rate megr* increases accordingly.

As shown in Fig. 19C, the correction coefficient at the time of increase of fuel supply amount k_{inc} is set to 5 1.0 when the increasing correction amount F_{inc} is zero, and becomes smaller as the increasing correction amount F_{inc} becomes larger. This correction coefficient at the time of increase of fuel supply amount k_{inc} is stored in the ROM 32 in advance in the form of a map shown in Fig. 10 19C.

This allows that the *EGR control valve passing-through gas flow rate megr* is calculated with higher accuracy.

Alternatively, the exhaust temperature T_e when the 15 retardation correction of ignition timing or the increasing correction of fuel supply amount is not in process may be obtained in advance as a function of the engine operating condition such as the engine speed NE and the required load L , the actual exhaust temperature 20 T_e may be detected or estimated and, then, the *EGR control valve passing-through gas flow rate megr* may be corrected based on the difference between the exhaust temperature T_e when the retardation correction of ignition timing or the increasing correction of fuel supply amount is not in process and the actual exhaust 25 temperature T_e . The same may be applied to the *cylinder-charged EGR gas amount Mcegr* in the engine steady operation and, thus, an explanation therefor is omitted.

In the internal combustion engine shown in Fig. 1, 30 as described above, the EGR supply pipe 21 downstream of the EGR control valve 22 is forked into the branches connected to the respective intake branches 12. In this configuration, in order to suppress unevenness of the amount of the EGR gas supplied to each cylinder, a choke 35 23 may be provided in each of the branches of the EGR supply pipe 21, as shown in Fig. 20.

In this case, first, in the engine steady operation,

5 a choke passing-through gas flow rate m_{ch} (g/sec), which is an amount of the EGR gas passing through the chokes 23, coincides with the EGR control valve passing-through gas flow rate m_{egr} . Therefore, as understood from the foregoing description, the choke passing-through gas flow rate m_{ch} in the engine steady operation can be calculated based on the difference ΔKL . Note that the choke passing-through gas flow rate m_{ch} represents a flow rate of the EGR gas flowing into the intake pipe.

10 On the other hand, in the engine transient operation, the choke passing-through gas flow rate m_{ch} does not always coincide with the EGR control valve passing-through gas flow rate m_{egr} . However, when an internal volume of the EGR supply pipe 21 from the EGR control valve 22 to the chokes 23 is relatively small, 15 m_{ch} substantially coincides with m_{egr} . Therefore, when the internal volume of the EGR supply pipe 21 from the EGR control valve 22 to the chokes 23 is relatively small, the choke passing-through gas flow rate m_{ch} can be calculated based on the difference ΔKL , either in the 20 engine steady or transient operation.

Figs. 21A, 21B and 21C show different internal combustion engines to which the present invention can be applied.

25 In an internal combustion engine shown in Fig. 21A, additional surge tanks 25 are connected to the intake branches 12 of each cylinder via respective intake pipe length control valves 24a.

30 The intake pipe length control valves 24a are closed when the engine speed is low, and are opened when the engine speed is high, for example. When the intake pipe length control valves 24a are closed, communication between the intake branches 12 and the additional surge tanks 25 is blocked to extend the effective length of the 35 intake pipe. In contrast, when the intake pipe length control valves 24a are opened, the intake branches 12 and

the additional surge tanks 25 are communicated with each other to shorten the effective length of the intake pipe. As a result, an efficient intake air pulsation is achieved, irrespective of the engine speed NE.

5 On the other hand, in an internal combustion engine shown in Fig. 21B, each of the intake branches 12 of the cylinders is provided with a respective pair of intake passages 12a and 12b therein separated by a respective partition wall 26, and each of the intake passages 12a and 12b is connected to the respective intake port 7. A swirl control valve 24b is disposed in one intake passage 12a of the pair of the intake passages 12a and 12b.

10 The swirl control valves 24b are closed when the engine load is low, and are opened when the engine load is high, for example. When the swirl control valves 24b are closed, the gas mixture flows into the cylinder only from the other intake passage 12b to create a swirl in the cylinder about the cylinder axis. In contrast, when the swirl control valves 24b are opened, the gas mixture flows into the cylinder from both intake passages 12a and 12b and, thus, a sufficient amount of fresh air is supplied to the cylinder.

15 In an internal combustion engine shown in Fig. 21C, a tumble control valve 24c is disposed at the bottom of internal space of each intake branch 12 of the cylinder.

20 The tumble control valves 24c are closed when the engine load is low, and are opened when the engine load is high, for example. When the tumble control valves 24c are closed, the gas mixture proceeds along the top of the internal wall of the intake branch 12, flows in the cylinder through a portion at the side of the exhaust valve 8 of the opening formed around the intake valve 7, falls down along the internal wall of the cylinder bore beneath the intake valve 8, proceeds on the top surface of the piston and, then, ascends along the internal wall of the cylinder bore beneath the intake valve 7, to create a swirl or tumble flow in the cylinder. In

contrast, when the tumble control valves 24c are opened, the gas mixture flows in the cylinder through the entire intake branch 12 and, thus, a sufficient amount of fresh air is supplied to the cylinder.

5 If a device controlling the air flow in the intake pipe, such as the intake pipe length control valve 24a, the swirl control valve 24b and the tumble control valve 24c, is referred to as an intake control valve, the engine load ratio KL may vary depending on whether the
10 intake control valve is closed or opened.

Therefore, in each internal combustion engine shown in Figs. 21A through 21C, maps representing the parameters a^* , b^* , c^* , e^* , r^* , h^* and i^* when the intake control valve is closed and maps representing these 15 parameters when the intake control valve is opened are obtained and stored, in advance, and the parameters may be calculated from either map depending on the condition of the intake control valve.

When the opening degree of the intake control valve 20 is controlled in a multi-step manner, each parameter may be set as a function of the opening degree of the intake control valve. More specifically, with regard to the parameter a^* , for example, a^* may be stored as a function of the engine speed NE , the opening degree of the EGR control valve STP and the opening degree of the intake 25 control valve in the form of a three-dimensional map. The same can be applied to the other parameters.

Therefore, generally speaking, it can be said that a 30 linear function expression of the intake pipe pressure P_m expressing the engine load ratios KL_{on} and KL_{off} or the difference ΔKL for different opening degrees of the intake control valve are obtained and stored in advance.

In this way, in each internal combustion engine 35 shown in Figs. 21A through 21C, each parameter is calculated using the map for the intake control valve being closed when the intake control valve is closed, and using the map for the intake control valve being opened

when the intake control valve is opened and, therefore, the engine load ratios K_{Lon} or K_{Loff} or the difference ΔKL is calculated accurately.

5 However, in the internal combustion engine of Fig. 21B, for example, the swirl flow is not created soon after the swirl control valve 24b is closed, but is created gradually with the elapse of time. This means that the engine load ratios K_{Lon} and K_{Loff} or the difference ΔKL cannot always be calculated accurately 10 even if the map used for calculating the parameter is switched in response to the switching of the swirl control valve 24b from the closed state to the opened state. The same can be applied to a case when the swirl control valve is turned open.

15 Therefore, it is preferable to change each parameter gradually with a predetermined changing rate, from the value calculated using the map for the condition of the intake control valve before the switching, to that calculated using the map for the condition of the intake 20 control valve after the switching. Accordingly, this allows that the engine load ratios K_{Lon} and K_{Loff} or the difference ΔKL is calculated accurately even during the switching of the intake control valve. Further, the 25 changing rate when the intake control valve is switched from the closed state to the opened state and that when it is switched from the opened state to the closed state, may be different from each other.

30 In the embodiments according to the present invention described above, the engine load ratios K_{Lon} and K_{Loff} or the difference ΔKL are calculated from the 35 intake pipe pressure P_m detected by the pressure sensor 39. Alternatively, the intake pipe pressure P_m may be estimated based on the opening degree of the throttle valve or an output of an air flow meter disposed in the intake duct 14 upstream of the throttle valve 17, for example, and the engine load ratio KL may be calculated

from the estimated intake pipe pressure P_m . Further alternatively, the intake pipe pressure P_m may be estimated using a computation model, for example, and the engine load ratio K_L may be calculated from the estimated intake pipe pressure P_m .

When the intake pipe pressure P_m is estimated based on the opening degree of the throttle valve, the intake pipe pressure P_m is obtained in advance as a function of the opening degree of the throttle valve, the engine speed N_E and the opening degree of the EGR control valve STP , and is stored in the form of a map.

10 The thus obtained P_m is read in in step 100 shown in Fig. 14 or in step 120 shown in Fig. 17.

15 On the other hand, if the intake pipe pressure P_m is estimated based on the output of the air flow meter, the estimated intake pipe pressure P_m may exceed the maximum pressure P_{mmax} which the intake pipe pressure P_m can accept, due to detection accuracy of the air flow meter or the like. However, as shown in Fig. 22B, in the range 20 of $P_m > P_{mmax}$, the engine load ratio K_{Lon} expressed with the above-described equation (2) may be larger than the engine load ratio K_{loff} expressed with the equation (3) and, in this case, the difference ΔK_L will be a negative value. Thus, if the estimated intake pipe pressure P_m exceeds the maximum pressure P_{mmax} , the difference ΔK_L 25 may not be calculated accurately.

Accordingly, as shown in Fig. 22A, in the range of $P_m > P_{mmax}$, maintaining of the difference ΔK_L at a constant value ΔK_{LC} , will eliminate such a problem.

30 Namely, the difference ΔK_L can still be calculated accurately when the estimated intake pipe pressure P_m exceeds the maximum pressure P_{mmax} .

35 Fig. 23 shows a calculation routine for the EGR control valve passing-through gas flow rate megr according to the embodiment explained referring to Figs.

22A and 22B. This routine is executed by interruption every predetermined time.

Referring to Fig. 23, first, in step 140, the intake pipe pressure P_m , the engine speed NE and the opening degree of the EGR control valve STP are read in. In the following step 141, the atmospheric temperature correction coefficient k_{tha} and the atmospheric pressure correction coefficient k_{pa} are calculated. In the following step 142, the intake pipe pressure b^* and engine load ratio c^* and r^* at the connecting point CP under the reference environmental condition are calculated from the maps of Figs. 7, 8B, and 12. In the following step 143, the parameters b , c and r are calculated by correcting b^* , c^* and r^* using k_{tha} and k_{pa} . In the following step 144, it is judged whether the obtained intake pipe pressure P_m is not higher than the maximum pressure P_{mmax} . If $P_m \leq P_{mmax}$, the process proceeds to step 145, where it is judged whether the obtained intake pipe pressure P_m is not higher than the intake pipe pressure b at the connecting point. If $P_m \leq b$, the process proceeds to step 146, where $a1^*$ and $e1^*$ are calculated from the maps of Figs. 6C and 11A. In the following step 147, the gradients a^* and e^* are set to $a1^*$ and $e1^*$, respectively. Then, the process proceeds to step 150. In contrast, if $P_m > b$ in step 145, the process proceeds to step 148, where $a2^*$ and $e2^*$ are calculated from the maps of Figs. 6D and 11B. In the following step 149, the gradients a^* and e^* are set to $a2^*$ and $e2^*$, respectively. Then, the process proceeds to step 150.

In the step 150, the parameters a and e are calculated by correcting a^* and e^* using k_{tha} and k_{pa} . In the following step 151, the engine load ratio $KLoft$ is calculated based on the equation (3) ($KLoft = a \cdot (P_m - b) + c$). In the following step 152, the engine load ratio $KLon$ is calculated based on the equation (2) ($KLon =$

$e \cdot (P_m - b) + r$. In the following step 153, the difference ΔK_L is calculated ($\Delta K_L = K_{Loff} - K_{Lon}$). Then, the process proceeds to step 155.

5 If $P_m > P_{mmax}$ in step 144, the process proceeds to step 154, where the difference is set to the constant ΔK_{LC} and, then, the process proceeds to step 155.

In step 155, the EGR control valve passing-through gas flow rate m_{egr} is calculated based on the equation (5) ($m_{egr} = k_{egr2} \cdot \Delta K_L$).

10 According to the present invention, it is possible to provide an apparatus for calculating an amount of a recirculated exhaust gas for an internal combustion engine which can obtain the exhaust gas recirculation control valve passing-through gas amount, simply and accurately.

15 While the invention has been described by reference to specific embodiments chosen for purposes of illustration, it should be apparent that numerous modifications could be made thereto by those skilled in the art without departing from the basic concept and scope of the invention.